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Two-Phase Heat Transfer of Low GWP Ternary Mixtures

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TWO-PHASE HEAT TRANSFER OF LOW GWP TERNARY MIXTURES

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ABSTRACT

Refrigerant blends obtained mixing hydrofluorocarbons (HFC) and hydrofluoroolefins (HFO) have recently been proposed as substitutes for high GWP (Global Warming Potential) fluids employed in refrigeration and air-conditioning systems. As a general trend, the dimension of pipes used in heat exchangers is decreasing: diameters around 5 mm are often employed in finned-tube coil heat exchangers and minichannels heat exchangers (with internal diameter around 1-2 mm) are also a common solution for the automotive sector and for air-cooled chillers.
Condensation and flow boiling heat transfer coefficients of zeotropic ternary mixtures R455A (R32, R1234yf and R744 at 21.5/75.5/3.0% by mass composition) and R452B (R32, R1234yf and R125 at 67.0/26.0/7.0% by mass composition) have been measured inside a minichannel (0.96 mm diameter) and inside a conventional tube (8.0 mm diameter). R455A exhibits a temperature glide around 10 K at 35 °C bubble temperature whereas R452B presents a temperature glide around 1 K at 40 °C bubble temperature. The experimental results are compared with selected correlations for condensation and flow boiling heat transfer which account for the additional mass transfer resistance occurring during two-phase heat transfer of zeotropic mixtures. It emerges the importance of including the mass transfer resistance for the prediction of heat transfer coefficient when considering high temperature-glide mixtures.

1. INTRODUCTION

European and international organizations have recently taken actions for phase-out of high Global Warming Potential (GWP) refrigerants. Refrigeration and air-conditioning industry is called to replace traditionally employed refrigerants while maintaining systems efficiency: the employment of low-GWP fluids represents a fundamental step to limit the carbon dioxide emissions of the installations. The search for alternative refrigerants is focused primarily on the use of natural fluids and on the employment of new synthetic refrigerants having low-GWP. Natural refrigerants are suggested as good alternatives to HFCs (hydrofluorocarbons) due to their thermodynamic properties and very low GWP; however, they are often flammable or toxic and cannot be considered as drop-in replacement fluids. Hydrofluoolefins (HFOs) are employed for replacing R134a in automotive air-conditioning because of their low GWP value, reduced atmospheric lifetime and comparable thermodynamic performance. Unfortunately, single-component low-GWP refrigerants cannot cover all the applications (e.g. drop-in refrigerants for R410A and R404A) and some zeotropic blends of HFOs and HFCs have been proposed. Zeotropic mixtures introduce an additional mass transfer resistance that leads to a degradation of the heat transfer performance and thus models developed for pure fluids cannot be directly applied. Therefore, it is important to collect new sets of data with these new mixtures that can be used for the assessment of the available two-phase heat transfer correlations (for both condensation and flow boiling) or for the development of new models.
Although several experimental and numerical investigations have been conducted on ternary non-azeotropic HFCs mixtures (Sami and Grell, 2000; Cavallini et al., 1997), relatively poor studies have been published on condensation and flow boiling performance of ternary HFOs/HFCs blends. Kondou et al. (2015) measured the heat transfer coefficients and pressure drop of R744/R32/R1234ze(E) mixtures at different mass compositions during condensation and evaporation inside a horizontal microfin tube with equivalent inner diameter of 5.35 mm. Mastrullo (2019) et al. and Lillo et al. (2019) respectively investigated the flow boiling heat transfer performance of ternary blends R452A and R488A inside a 6.0 mm inner diameter channel.

The present study is aimed at investigating the condensation and flow boiling heat transfer performance of two ternary low-GWP non-azeotropic mixtures, named R455A and R452B, inside a 8.0 mm ID tube and a 0.96 mm minichannel.

2. EXPERIMENTAL APPARATUS AND DATA REDUCTION

Experimental tests have been performed with two different test sections consisting of two tube-in-tube heat exchangers with internal diameter equal to 8.0 mm and 0.96 mm. Each test section is installed in a dedicated experimental test rig. During condensation tests, the subcooled refrigerant is sent by a magnetic-driven gear pump to a tube-in-tube evaporator where it is vaporized and superheated. The superheated vapor then passes through a pre-conditioning sector where it partially condenses reaching the desired vapor quality conditions at the inlet of the measuring sector. Water is employed as secondary fluid to cool down the refrigerant in the measuring sector and is provided at different temperatures by thermal baths and electrical heaters. Differently, when flow boiling tests are performed, the subcooled refrigerant is sent by the pump directly to the test section where it is heated and vaporized using hot water. After the measuring sector, during both condensation and flow boiling tests, the refrigerant is collected in a post-condenser where it is fully condensed and subcooled.
In the 8.0 mm diameter test section, the quasi-local heat transfer coefficient is obtained by measuring the heat flow rate Q exchanged on the water side over the surface area A:

$$\mathrm{HTC} = \frac{Q}{A \cdot \Delta T_{LMTD}} = \frac{\dot{m}_w \cdot c_{p,w} \cdot \left(T_{w,out} - T_{w,in}\right)}{A \cdot \Delta T_{LMTD}} \tag{1}$$

where the heat flow rate Q is determined from the measured water mass flow rate $\dot{m}_w$ and water temperatures at the inlet $T_{w,in}$ and outlet $T_{w,out}$ of the measuring sector. The logarithmic mean temperature difference is calculated using the saturation temperature T_{sat} and the wall temperatures T_{wall} measured at the inlet and outlet of the measuring sector:

$$\Delta T_{LMTD} = \frac{(T_{sat,in} - T_{wall,in}) - (T_{sat,out} - T_{wall,out})}{\ln\left(\dfrac{T_{sat,in} - T_{wall,in}}{T_{sat,out} - T_{wall,out}}\right)} \tag{2}$$

The saturation and wall temperatures are measured by T-type thermocouples. In particular, the wall temperatures at the inlet and outlet of the test section are obtained as the mean value of the readings of four thermocouples, inserted into four equidistant axial grooves around the perimeter of the test tube. The inlet and outlet vapor quality in the measuring sector are computed using NIST Refprop 10.0 (Lemmon et al., 2010), given the measured pressure, the evaluated enthalpy (from an energy balance on the water side) and the nominal composition of the mixture. Detailed description of the experimental apparatus and data reduction procedure can be found in Del Col (2010).
The 0.96 mm diameter test section is equipped with thermocouples installed both on the copper wall and on the water path. The water thermocouples are used to determine the water temperature profile along the test section, which is in turn used to obtain the local heat flux q' and the local heat transfer coefficient:

$$q'(z) = -\dot{m}_w \cdot c_{p,w} \cdot \frac{\mathrm{d}T_w(z)}{\mathrm{d}z} \cdot \frac{1}{\pi d} \tag{3}$$

$$\mathrm{HTC}(z) = \frac{q'(z)}{T_{sat}(z) - T_{wall}(z)} \tag{4}$$

where d is the inner diameter of the tube and z is the axial position. The saturation temperature in the measuring sector is obtained from the measured pressure, the specific enthalpy (determined from an energy balance on the water side) and the mass composition of the mixture. The local wall temperatures are determined from the measurements of thirteen T-type thermocouples embedded in the channel wall. The local vapor quality is calculated using the local pressure, the local enthalpy and the mixture composition. More details about the data reduction in the 0.96 mm test section are reported in Del Col et al. (2017) and Azzolin et al. (2019).
The uncertainty analysis is carried out following the JCGM (2008) guidelines. For condensation tests, the average expanded uncertainty on the heat transfer coefficient (with 95% confidence level) is 4.4% for the 8.0 mm diameter channel and 6.1% for the 0.96 mm minichannel. The average expanded uncertainty on the vapor quality is equal to ±0.01 and ±0.02 for the 8.0 mm and the 0.96 mm channel, respectively. For flow boiling tests, the average expanded

uncertainty on the heat transfer coefficient and vapor quality is respectively equal to 5.2% and ±0.01 for the 8.0 mm channel, while for the 0.96 mm minichannel it corresponds to 7.9% and ±0.03.
In the present work, ternary non-azeotropic blends R455A (R1234yf/R32/R744 at 75.5/21.5/3% by mass composition) and R452B (R1234yf/R32/R125 at 26/67/7%) are tested during condensation and flow boiling. Mixture R455A can be employed in commercial refrigeration for low temperature applications, while R452B is suitable for air-conditioning and heat pumps applications. In Table 1, some thermodynamic and transport properties of the tested blends are reported, compared to those of the pure fluid components R32, R1234yf and R125. Properties are computed using NIST Refprop 10.0 (Lemmon et al., 2010).

Table 1: Properties of tested mixtures and pure components at 40 °C and 18.2 °C mean saturation temperature computed using NIST Refprop 10.0 (Lemmon et al., 2010).

	R32	**R1234yf**	**R125**	**R455A**	**R452B**
$GWP_{100\text{-years}}$	677	<1	3170	146	676
Mean saturation temperature equal to 40 °C					
p_{sat} [bar]	24.78	10.18	20.08	17.62	22.87
ρ_l [kg m^{-3}]	893.0	1033.8	1088.4	988.9	924.7
ρ_g [kg m^{-3}]	73.3	57.8	142.5	61.0	79.8
T_{dew} [°C]	-	-	-	44.9	40.6
T_{bubble} [°C]	-	-	-	35.1	39.4
Mean saturation temperature equal to 18.2 °C					
p_{sat} [bar]	14.02	5.61	11.47	10.07	12.97
ρ_l [kg m^{-3}]	988.5	1116.2	1228.3	1082.7	1024.7
ρ_g [kg m^{-3}]	38.7	31.1	73.9	31.5	42.1
T_{dew} [°C]	-	-	-	23.8	18.8
T_{bubble} [°C]	-	-	-	12.6	17.6

3. CONDENSATION HEAT TRANSFER RESULTS

Condensation tests have been performed with R455A and R452B at mean saturation temperature (between dew and bubble point) equal to 40 °C and mass velocity equal to 200, 400 and 600 kg m^{-2} s^{-1}. The vapor quality change in the 8.0 mm test section is always lower than 0.2 to allow the determination of a quasi-local heat transfer coefficient while, in the 0.96 mm diameter tube, the local vapor quality is experimentally determined.
In Fig. 1 the experimental heat transfer coefficients of R455A inside the 8.0 mm and the 0.96 mm diameter channels are reported. The heat transfer coefficient is found to increase with mass velocity and vapor quality. Higher heat transfer coefficients can be obtained in the 0.96 mm diameter minichannel compared to the 8.0 mm channel at mass velocity equal to 400 kg m^{-2} s^{-1} and 600 kg m^{-2} s^{-1}. In particular, at G = 600 kg m^{-2} s^{-1} the heat transfer coefficient is reduced by 30% when the channel diameter is increased from 0.96 mm to 8.0 mm. This is in agreement with the Cavallini et al. (2006) correlation for annular flow, for which the heat transfer coefficient is a function of the hydraulic diameter to the -0.2 power. In fact, applying the Cavallini et al. (2006) correlation for annular flow, the heat transfer coefficient inside the larger diameter channel should be 35% lower than that obtained in the minichannel. Instead, considering the lowest value of mass velocity (G = 200 kg m^{-2} s^{-1}), the heat transfer coefficient in the 8.0 mm channel is reduced only by 6% with respect to the 0.96 mm diameter minichannel. The more evident effect of the channel diameter observed at the highest value of mass velocity is due to the different flow patterns inside the two channels: indeed, the hypothesis of annular flow may be satisfied at G = 600 kg m^{-2} s^{-1} for both the channels, but not at G = 200 kg m^{-2} s^{-1} where the flow pattern is expected to be stratified or stratified-wavy in the 8.0 mm channel.

3.1 Assessment of condensation correlations

The design of heat exchangers with non-azeotropic mixtures as working fluids requires precise correlations that must be able to consider the effect of the additional mass transfer resistance due to the different volatility of the components. The approach proposed in the present work applies the equilibrium method which was first introduced by Silver (1947) and Bell and Ghaly (1973). This method assumes complete mixing and overall equilibrium in the liquid and vapor phases. The sensible heat is removed from the vapor and transferred to the interface by convection

with a heat transfer coefficient evaluated under the hypothesis that only vapor is present using the mass flux and properties of the vapor. The condensation heat transfer coefficient of the mixture can be determined by correcting the film heat transfer coefficient calculated with a model for pure fluid through an additional heat transfer resistance in the vapor.

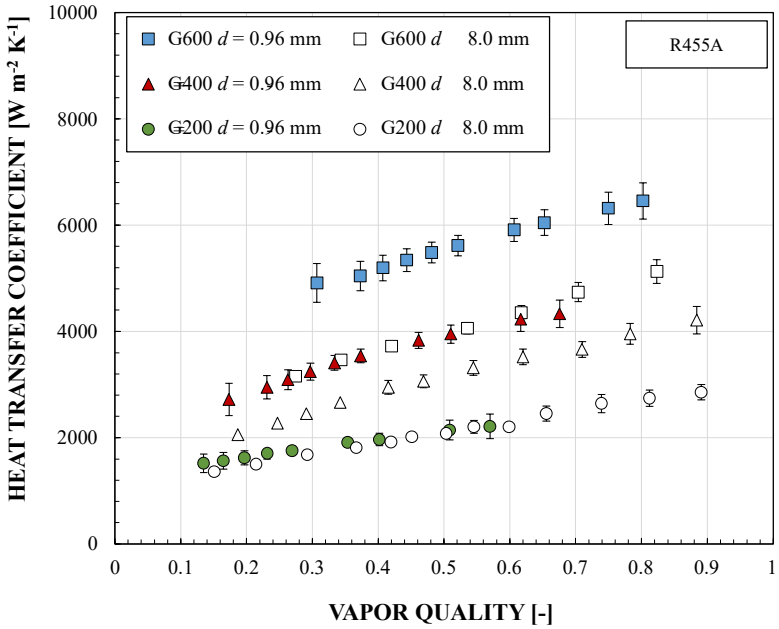

Figure 1: Experimental heat transfer coefficient of R455A versus vapor quality at mass velocity G equal to 200, 400 and 600 kg m^{-2} s^{-1} measured in the 0.96 mm and in the 8.0 mm internal diameter channel.

With this approach, the overall heat transfer coefficient of the mixture HTC_{mix} can be expressed as follows:

$$\mathrm{HTC}_{mix} = \left(\frac{1}{\mathrm{HTC}_f} + \frac{\partial Q_{sg} / \partial Q_t}{\mathrm{HTC}_g} \right)^{-1} = \left[\frac{1}{\mathrm{HTC}_f} + \frac{x \cdot c_{p,g} \cdot \left(\Delta T_{GL} / \Delta h_{mix} \right)}{\mathrm{HTC}_g} \right]^{-1} \tag{5}$$

where HTC_f is the liquid film heat transfer coefficient computed with a condensation model for pure fluids using the thermodynamic and transport properties of the mixture, HTC_g is the heat transfer coefficient of the vapor phase flowing alone in the channel evaluated using the Dittus-Boelter correlation, $\partial Q_{sg}/\partial Q_t$ is the ratio between the sensible heat flow rate removed from the vapor and the total heat duty, x is the vapor quality, $c_{p,g}$ is the specific heat of the vapor, ΔT_{GL} is the temperature glide of the mixture and Δh_{mix} is the isobaric enthalpy change of the mixture.

Fig. 2a) shows the ratio between the calculated condensation heat transfer coefficient of the mixture ($\mathrm{HTC}_{mix,CALC}$) and the measured heat transfer coefficient ($\mathrm{HTC}_{mix,EXP}$) in the 0.96 mm diameter channel. The predicted heat transfer coefficient for the mixture ($\mathrm{HTC}_{mix,EXP}$) has been calculated considering the Cavallini et al. (2006) model for condensation of pure fluids coupled with the SBG (Silver, Bell and Ghaly) correction term for non-azeotropic mixtures. The Cavallini et al. (2006) model (with the SBG correction) is able to predict the present experimental results in the 0.96 mm channel with mean absolute deviation e_{AB} = 2.5% and standard deviation σ_N = 2.7% for R455A, e_{AB} = 15.8% and σ_N = 8.8% for R452B.

Fig. 2b) reports the predicted heat transfer coefficients of R455A and R452B in the 8.0 mm channel using the Del Col et al. (2005) model, developed for zeotropic mixtures. The experimental heat transfer coefficients in the 8.0 mm tube are predicted by the Del Col et al. (2005) model with e_{AB} = 14.7% and σ_N = 6.8% for R455A and e_{AB} = 10.4% and σ_N = 12.3% for R452B. The condensation heat transfer coefficients inside the 8.0 mm channel are underestimated by the Del Col et al. (2005) model at low mass velocity. The deviation observed at G = 200 kg m^{-2} s^{-1} may be related to flow patterns and liquid stratification.

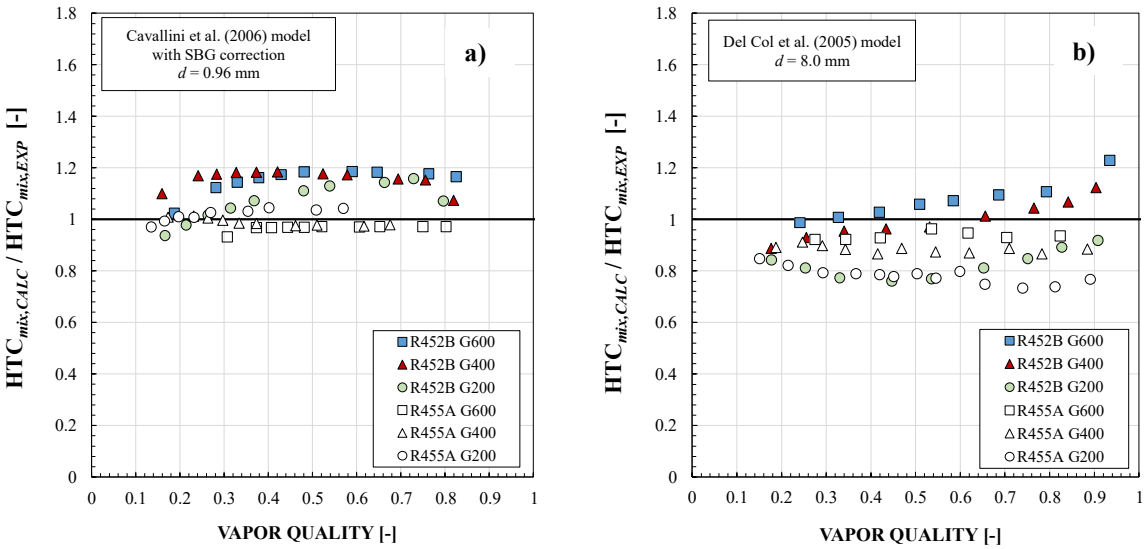

Figure 2: Comparison between predicted and experimental heat transfer coefficients of R455A and R452B using the Cavallini et al. (2006) model corrected for non-azeotropic mixtures with the SBG correction in the 0.96 mm channel (a) and the Del Col et al. (2005) model for zeotropic mixtures in the 8.0 mm channel (b) (*G* is the mass velocity).

4. PERFORMANCE EVALUATION CRITERION DURING CONDENSATION

The comparison among different refrigerants during convective condensation should be based not only on the heat transfer performance but also on the pressure drop, which affects the refrigerant saturation temperature and thus the driving temperature difference between saturation and wall. In order to allow the performance analysis of different refrigerant candidates for condenser applications, a Performance Evaluation Criterion (PEC) is here adopted, referring to the analysis performed by Cavallini et al. (2010). Cavallini et al. (2010) introduced the Penalty Factor (*PF*) which is defined as the product of the saturation pressure decrease due to pressure drop (ΔT_{sat}) and the saturation-minus-wall temperature difference (ΔT_{dr}), which is the driving temperature difference for the heat transfer process:

$$PF = \Delta T_{sat} \cdot \Delta T_{dr} \tag{6}$$

Cavallini et al. (2010) proved that both the multiplying terms in Eq. (6) are associated with exergy losses which penalize the cycle efficiency and affect the compressor power consumption. As reported in Cavallini et al. (2010), the *PF* factor can be calculated with the following equation:

$$PF = \frac{G \cdot d \cdot T_{sat}}{4 \cdot \mathrm{HTC}} \cdot \left(\frac{1}{\rho_l} - \frac{1}{\rho_g} \right) \cdot \frac{\partial p_{fr}}{\partial z} \tag{7}$$

where $\partial p_{fr}/\partial z$ is the frictional pressure drop along the tube length. As shown by Eq. (7), the application of this methodology requires the knowledge of both the heat transfer coefficient and the frictional pressure gradient at given operating conditions. It may be underlined that the Penalty Factor combines the two exergy losses terms in order to have a direct comparison between two different refrigerants at the same operative conditions. In particular, the *PF* factor can be assumed as a quantitative criterion to rank the heat transfer performance of different working fluids: the smaller the *PF*, the better is the performance potential of the refrigerant. This method is applied to mixtures R455A and R452B, pure fluid components, and blends R410A and R404A in the 0.96 mm channel. Imposing a saturation temperature for pure fluids and a mean saturation temperature for mixtures both equal to 40 °C, the Penalty Factor is evaluated for a constant value of mass velocity G = 400 kg m^{-2} s^{-1}. The wall temperature is kept 10 K lower than the saturation temperature. The models of Cavallini et al. (2006) and Friedel (1979) are respectively employed for the prediction of the heat transfer coefficient and the frictional pressure drop of the considered fluids. The results of this analysis are reported in Fig. 3a): it can be noticed that the low GWP refrigerant R1234yf shows the highest PF in the whole vapor quality range, while mixture R410A shows the lowest PF value

(PF = 1.16 K^2 at x = 0.50). The curves pertaining to mixtures R455A and R452B stand between the pure fluids R32 and R1234yf and, as the R32 mass fraction increases, the PF decreases.
By looking at the PF definition, if the operative conditions are set to obtain the same PF, a different value of the mass velocity can be calculated for each fluid. Once the mass velocity values are identified, the condensation heat transfer coefficients can be calculated and compared. These mass velocities are the ones that lead the fluids to show the same exergy penalization during the condensation process. Fig. 3b) shows the heat transfer coefficients calculated using the correlation of Cavallini et al. (2006), corrected for mixtures when needed, for a specific mass velocity for each fluid, at PF = 5 K^2 and vapor quality x = 0.5. R32 reaches the highest heat transfer coefficient, while R455A and R1234yf show the lowest values. Mixture R455A is close to fluid R1234yf in terms of heat transfer coefficient, while the heat transfer performance of R452B is about 25% lower than that of R32. It is interesting to notice that the heat transfer performance of R452B is comparable to the one of R410A, thus confirming the suitability as drop-in replacement fluid. Instead, the heat transfer coefficients of R455A are 20-37% lower than those of R404A, which is favoured by the smaller temperature glide.

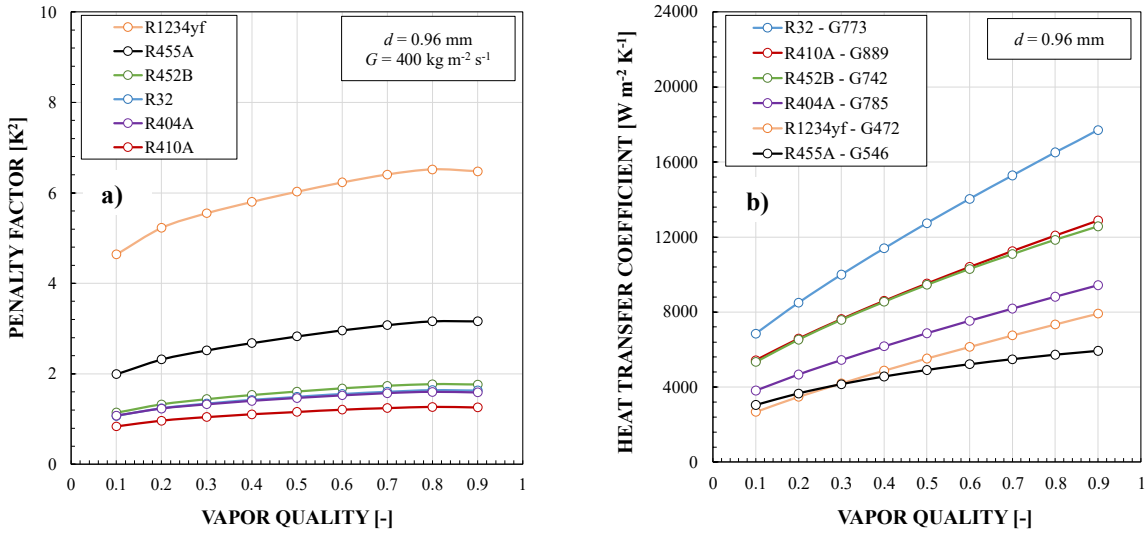

Figure 3: Performance evaluation analysis among R455A, R452B, R1234yf, R32, R404A and R410A inside the 0.96 mm channel. a) Penalty factor at G = 400 kg m^{-2} s^{-1}; b) heat transfer coefficients calculated using Cavallini et al. (2006) model, with the mass velocities evaluated at PF = 0.5 and x = 0.5 (G is the mass velocity).

5. FLOW BOILING HEAT TRANSFER RESULTS

The flow boiling heat transfer characteristics of R455A and R452B have been investigated in the 8.0 mm and the 0.96 mm diameter channel. The vapor quality change in the 8.0 mm test section is always smaller than 0.2.
Figs. 4 and 5 show the effect of mass velocity and heat flux on the flow boiling heat transfer coefficient of R455A and R452B in the 8.0 mm channel. In Figs. 4 and 5, the heat transfer coefficients of R455A and R452B are reported at mass flux equal to 200 and 400 kg m^{-2} s^{-1}, heat flux q' equal to 7 kW m^{-2} and 15.5 kW m^{-2} and 11 °C mean saturation temperature. The heat transfer coefficient of R452B is on average 60-70% higher than the one of R455A and this may be related to the properties of R452B (higher liquid thermal conductivity, higher reduced pressure) and to the smaller temperature glide. The temperature glide is related to the composition of the mixture and a mass transfer resistance develops in both the liquid and the vapor phases, leading to a heat transfer penalization in high-temperature glide blends compared to smaller-temperature glide mixtures (Azzolin et al., 2016). For both the fluids, at fixed vapor quality, the heat transfer coefficient is found to increase with heat flux and mass velocity. With the increasing mass velocity, the higher turbulence inside the liquid and the vapor phase and the reduced heat and the mass transfer resistances lead to a heat transfer enhancement. Furthermore, at fixed mass velocity, the heat transfer coefficient of both the blends increases with vapor quality up to 0.8 vapor quality (after which the heat transfer coefficient may decrease due to the occurrence of partial dryout). The increase of the heat transfer coefficient with vapor quality is more evident for R455A, as the heat transfer increase in the 0.1-0.85 vapor quality range is equal to 60% at q' = 7 kW m^{-2} while, for R452B, it corresponds to 15% at the same heat flux and vapor quality range. It

appears that the convective boiling component is more important for R455A with respect to R452B, even if the vapor density is the same. This may be linked to the higher temperature glide and to the lower reduced pressure, which penalize the nucleate boiling contribution in the case of R455A compared to R452B (Berto et al, 2020). Moreover, if considering the heat transfer coefficients measured at 15.5 kW m^{-2}, G = 200 kg m^{-2} s^{-1} and vapor quality x = 0.5, the heat transfer coefficient in the case of R455A is about 40% lower than the one measured with R452B and this difference cannot be explained accounting only for the thermal properties. The reason must be related to the mass transfer resistance which acts on both the nucleate boiling and the convective boiling contributions.

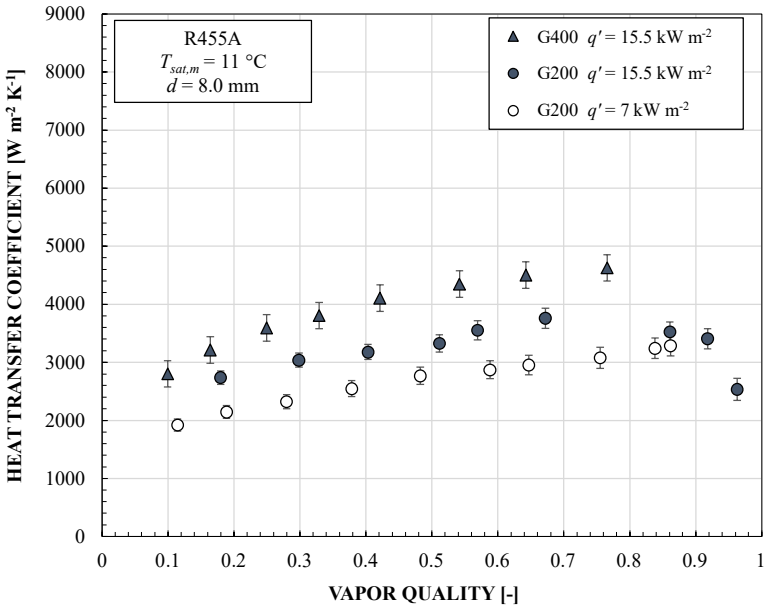

Figure 4: Effect of mass velocity and heat flux on the flow boiling heat transfer coefficient of R455A at 11 °C mean saturation temperature in the 8.0 mm diameter channel (G is the mass velocity).

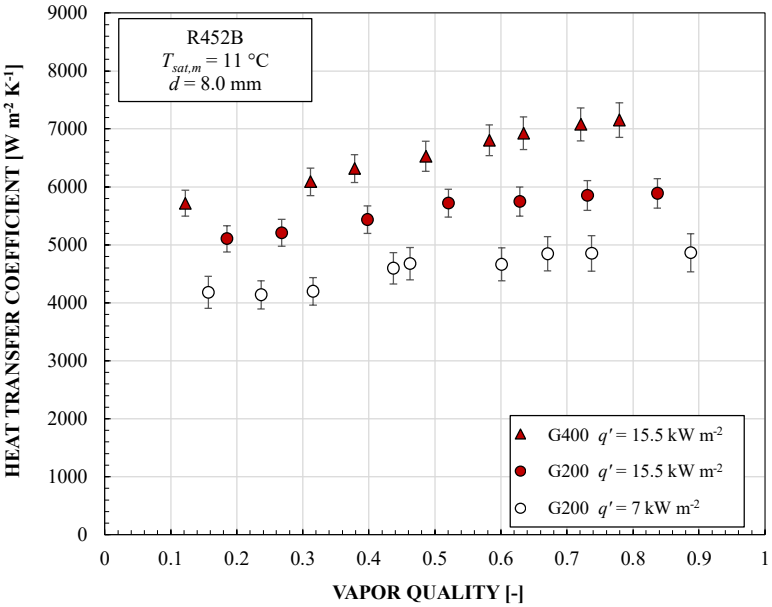

Figure 5: Effect of mass velocity and heat flux on the flow boiling heat transfer coefficient of R452B at 11 °C mean saturation temperature in the 8.0 mm diameter channel (G is the mass velocity).

5.1 Assessment of flow boiling correlations

A simple approach to predict the flow boiling heat transfer coefficient of zeotropic mixtures relies on the application of a specific correction term to heat transfer correlations developed for pure fluids in order to account for the additional mass transfer resistance. In particular, Shah (2015) proposed to modify the pure fluid correlations by using the Thome and Shakir (1987) correction factor in the nucleate boiling region and the Bell and Ghaly (1973) method in the convective region. The Thome and Shakir (1987) factor accounts for the fact that, during nucleate boiling, the more volatile component from the liquid bulk has to diffuse through the diffusion layer, which becomes richer in the less volatile component, thus increasing the bubble temperature. For example, when adopting the Gungor and Winterton (1986) or Wattelet et al. (1994) heat transfer correlations, the method proposed by Shah (2015) can be respectively applied as follows:

$$\mathrm{HTC}_{mix,GW} = F_c \cdot S_{GW} \cdot \mathrm{HTC}_{nb} + \left(\frac{1}{E_{GW} \cdot \mathrm{HTC}_l} + \frac{\partial Q_{sg} / \partial Q_t}{\mathrm{HTC}_g} \right)^{-1} \tag{8}$$

$$\mathrm{HTC}_{mix,WT} = \left[\left(F_c \cdot \mathrm{HTC}_{nb} \right)^{2.5} + \left(\frac{1}{E_{1,WT} \cdot E_{2,WT} \cdot \mathrm{HTC}_l} + \frac{\partial Q_{sg} / \partial Q_t}{\mathrm{HTC}_g} \right)^{-2.5} \right]^{1/2.5} \tag{9}$$

where F_c is the Thome and Shakir (1987) factor, S_{GW} is the suppression factor of the nucleate boiling mechanism defined by Gungor and Winterton (1986), HTC_{nb} is the heat transfer coefficient calculated with the Cooper (1984) pool boiling correlation, E_{GW}, $E_{1,WT}$ and $E_{2,WT}$ are enhancement factors for the convective boiling contribution, HTC_l and HTC_g are evaluated using the Dittus-Boelter correlation for the liquid and the vapor phases, respectively. Fig. 6a) shows the comparison between calculated and experimental heat transfer coefficients of R455A and R452B in the 8.0 mm channel, considering the predictions of the models by Gungor and Winterton (1986) and Wattelet et al. (1994) corrected for mixtures. The model of Wattelet et al. (1994), corrected for mixtures with Shah (2015) approach, is able to predict the heat transfer coefficient with better accuracy for both the blends (e_{AB} = 11.9% and σ_N = 5.1% for R455A, e_{AB} = 8.5% and σ_N = 9.9% for R452B). Instead, the Gungor and Winterton (1987) model, corrected with the Shah (2015) method, predicts the experimental results with e_{AB} = 15.9% and σ_N = 17.1% for R455A and e_{AB} = 9.4% and σ_N = 9.5% for R452B.

Fig. 6b) reports the comparison between the heat transfer coefficients of R455A calculated using the Sun and Mishima (2009) correlation and the experimental data in the 0.96 mm minichannel. Since in the Sun and Mishima (2009) correlation the convective boiling contribution is not considered, only the Thome and Shakir (1987) factor F_c for nucleate boiling suppression is applied in order to account for the mass transfer resistance:

$$\mathrm{HTC}_{mix,SM} = \frac{6 \cdot \mathrm{Re}_{lo}^{1.05} \cdot \left(F_c \cdot \mathrm{Bo} \right)^{0.54}}{\mathrm{We}_{lo}^{0.191} \cdot \left(\frac{\rho_l}{\rho_g} \right)^{0.142}} \cdot \frac{\lambda_l}{d} \tag{10}$$

where Bo is the boiling number (= $q'/(G\ \Delta h_{mix})$), Re_{lo} (= $G\ d\ /\mu_l$) and We_{lo} (= $G^2\ d\ /(\rho_l\ \sigma)$) are respectively the Reynolds number and the Weber number of the liquid flowing alone in the channel, λ_l is the liquid thermal conductivity, μ_l is the liquid dynamic viscosity, σ is the surface tension. In the work of Azzolin et al. (2016), the application of the Thome and Shakir (1987) factor to the boiling number Bo to account for the mass diffusion effect of the mixture was found to be necessary and appropriate, enabling accurate predictions of the heat transfer coefficient in minichannels. As shown by Fig. 6b), the Sun and Mishima (2009) model, corrected with the F_c factor, provides accurate predictions of the R455A data in the 0.96 mm diameter channel (e_{AB} = 4.8%, σ_N = 4.9%).

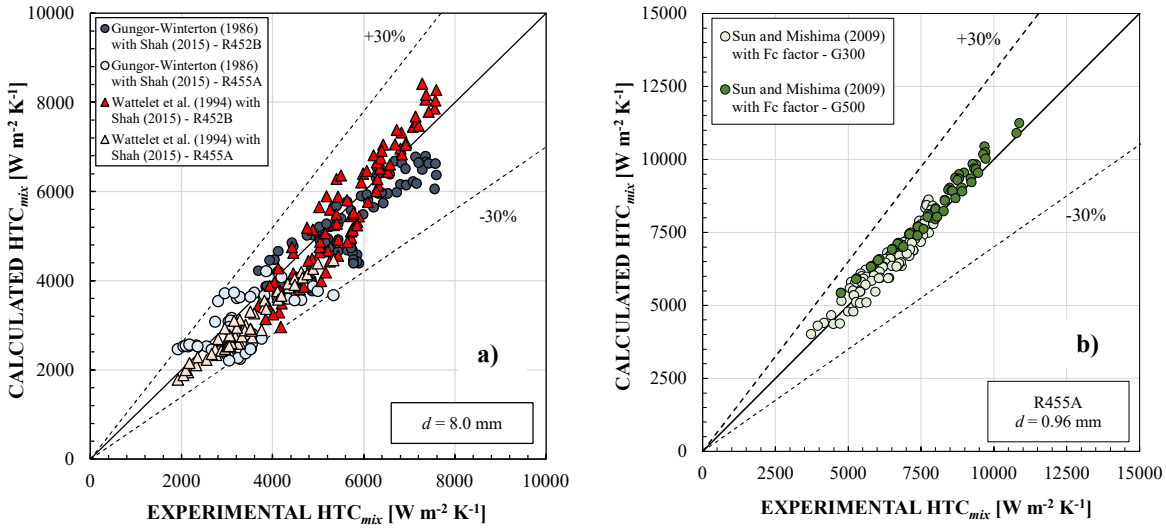

Figure 6: Comparison between predicted and experimental heat transfer coefficients: a) HTC_{mix} obtained using the Wattelet et al. (1994) and the Gungor and Winterton (1986) models with the Shah (2015) approach in the 8.0 mm channel; b) HTC_{mix} obtained using the Sun and Mishima (2009) model with F_c factor by Thome and Shakir (1987) in the 0.96 mm minichannel (G is the mass velocity).

6. CONCLUSIONS

Condensation and flow boiling heat transfer coefficients of ternary non-azeotropic mixtures in a 8.0 mm and a 0.96 mm diameter channel are reported in the present paper. The tested blends are R455A ($GWP_{100\text{-years}}$ = 146), ternary blend of R32, R1234yf and R744 at 21.5/75.5/3.0% by mass composition, and R452B ($GWP_{100\text{-years}}$ = 676), blend of R32, R1234yf and R125 at 67.0/26.0/7.05 by mass composition.

Heat transfer models developed for pure fluids need to be corrected for non-azeotropic mixtures as the phase change process is not isothermal and the change in the composition of the liquid/vapor phases is responsible for the formation of a mass transfer resistance. With respect to condensation tests, the best predictions for both the mixtures are provided by the Del Col et al. (2005) model for the 8.0 mm channel and by the Cavallini et al. (2006) model, corrected with the Silver (1947) and Bell and Ghaly (1973) approach, for the 0.96 mm minichannel. When flow boiling tests are considered, good accuracy is obtained using the Wattelet et al. (1987) model modified as proposed by Shah (2015) for the 8.0 mm channel, and applying the Sun and Mishima (2009) correlation with F_c factor defined by Thome and Shakir (1987) for the 0.96 mm minichannel.

A comparative analysis between mixtures and pure components during condensation that accounts for both the heat transfer coefficient and pressure drop is performed in the 0.96 mm minichannel by using the Penalty Factor (PF) as a performance evaluation criterion. The mass velocities that lead to the same exergy penalization are the lowest for R455A and R1234yf, which are found to be the most penalized fluids in terms of condensation heat transfer performance.

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